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09JUN03 E813252-1 D00393
P01/7700 0.00-0313134.9

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## Request for grant of a patent

*(See the notes on the back of this form. You can also get an explanatory leaflet from the Patent Office to help you fill in this form)*

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| | | |
|---|---|---|
| 1. | Your reference | 2003P08272 GB / / CF / GD |
| 2. | Patent application number *(The Patent Office will fill in this part)* | 0313134.9 — 7 JUN 2003 |
| 3. | Full name, address and postcode of the or of each applicant *(underline all surnames)* | DEMAG DELAVAL INDUSTRIAL TURBOMACHINERY LIMITED<br>Ruston House, Waterside South,<br>Lincoln, Lincolnshire LN5 7FD |
| | Patents ADP number *(if you know it)* | 8648602001 |
| | If the applicant is a corporate body, give the country/state of its incorporation | UNITED KINGDOM |
| 4. | Title of the invention | TILT PAD BEARING PIN |
| 5. | Name of your agent *(if you have one)*<br><br>"Address for service" in the United Kingdom to which all correspondence should be sent *(including the postcode)* | Siemens Plc<br>Intellectual Property Department<br>The Lodge, Roke Manor<br>Romsey, Hampshire SO51 0ZN |
| | Patents ADP number *(if you know it)* | 847219300 2 |
| | If you are declaring priority from one or more earlier patent applications, give the country and the date of filling of the or of each these earlier applications and *(if you know it)* the or each application number | Country / Priority application number *(if you know it)* / Date of filing *(day / month / year)* |
| | If this application is divided or otherwise derived from an earlier UK application, give the number and the filling date of the earlier application | |
| | Is a statement of inventorship and of right to grant of a patent required in support of this request? (Answer ,Yes' if: | Yes |

IDNR: 7100 / V: 02-1.00 / B: Val

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a) any application named in part 3 is not an inventor,or
b) there is an inventor who is not named as an applicant, or
c) any named applicant is a corporate body.
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| Abstract | 0 |
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| Priority documents | |
| Translation of priority documents | |
| Statement of inventorship and right to grant a patent *(Patents Form 7/77)* | 1 |
| Request for preliminary examination and search *(Patents Form 9/77)* | |
| Request for substantive examination *(Patents Form 10/77)* | |
| Any other documents *(please specify)* | Description includes drawings as graphs. |

11. I/We request the grant of a patent on the basis of this application

Signature: C French
Clive French
Intellectual Property Department

Date: 06.06.2003

12. Name and daytime telephone number of Person to contact in the United Kingdom

Clive French + 44 1794 83 3573

# Tilt pad bearing pin
# for
# gas turbine rotor shaft bearings

Background.

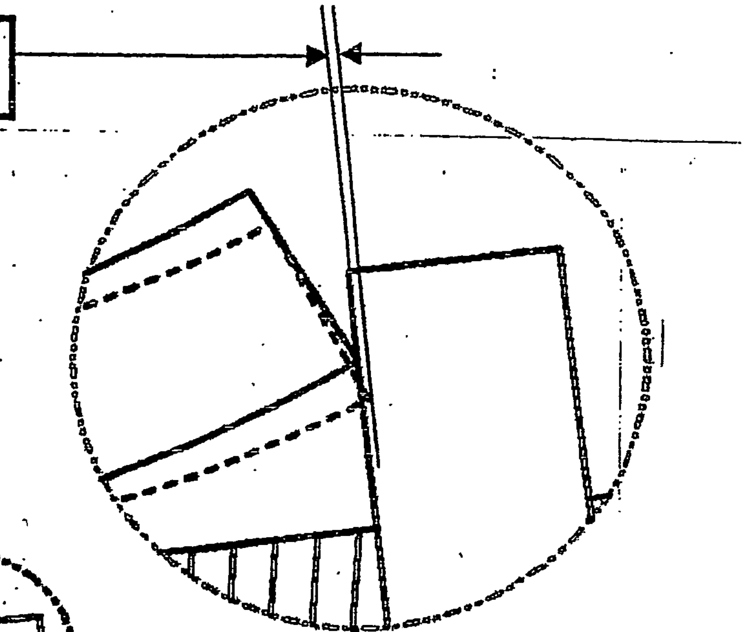
Gas turbine engine rotor shaft bearing assemblies may be of the plain bearing type which are segmented (radially self–aligning), retained in an outer carrier, oil pressure lubricated and known as tilt pad bearings. Such bearings are subject to high speed and load when the engine is running and very small out of limit wear in a bearing can have disastrous effects on an engine. It may cause engine shut-down which, for this kind of engine, is usually a very costly matter. For acceptable engine life it is therefore vital the bearings perform reliably.

It is typical in such segmented plain bearing arrangements to restrain radial movement of each segment by use of location pins which are fixed in the bearing assembly outer carrier and abut to segments either at segment end faces or at the side face of an internal hole or passage in the segment body. It is also typical the pin will be a parallel round pin throughout its length, but a known variation is for a pin with spheroid tip on a round pin stem, the sphere diameter being greater than the round pin stem.

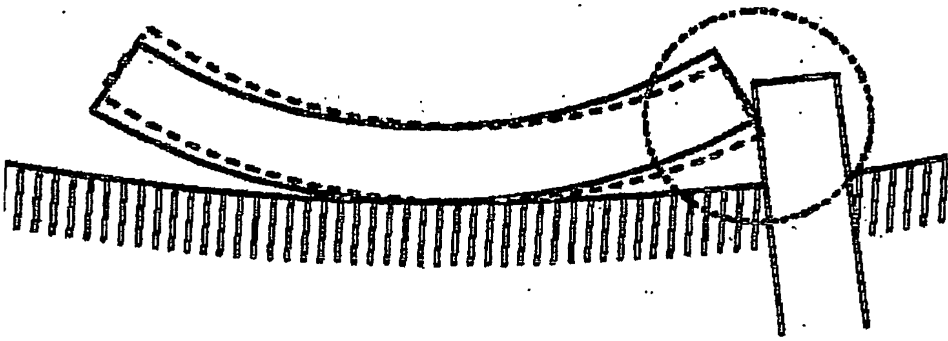
It has been found that at contact faces between the bearing segments and bearing outer carrier, fretting wear may occur which allows segments to move beyond design limits and thus cause bearing related engine problems. It is believed the fretting is caused by a small amount of sliding action under pressure between segment and carrier and that the action is made possible with prior art pin designs. The subject invention, a tapered pin, prevents the sliding action and thus there is no fretting wear.

# Sliding - Pads Restrained by End Pins

Sliding Distance



| Pad OD | Angle | Sliding Distance |
|---|---|---|
| 89.3mm | 0.1° | 8.7µm |
| 85mm | 0.1° | 8.7µm |



## Pad Sliding

Fretting is only possible with a sliding contact between metal surfaces. It is not possible if the motion of the pads in the carrier is purely rocking.

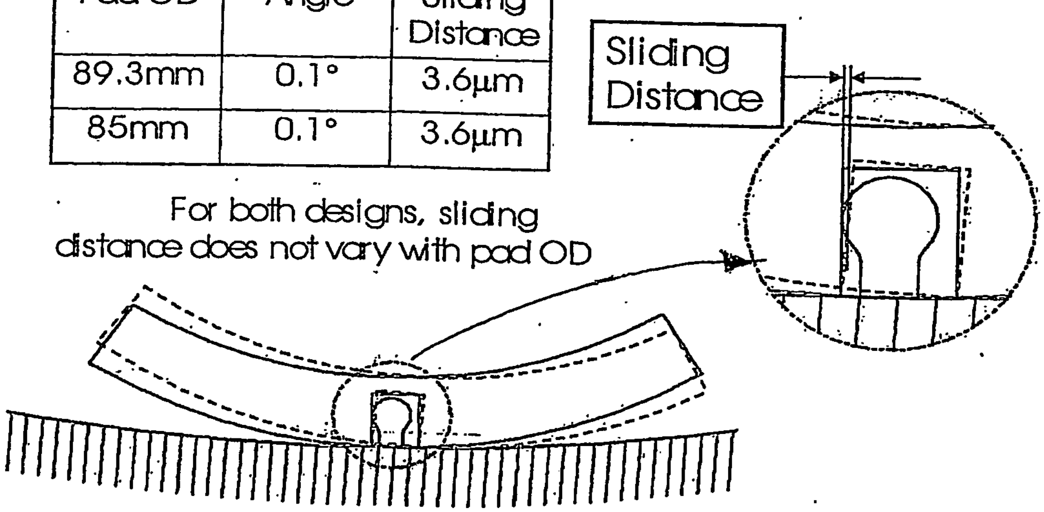
Sliding can occur in a circumferential direction with both Glacier and Sartorius bearings

If a tilt pad was able to rock freely, the edges of the pad would move radially and circumferentially as the pad rocks. Glacier bearing pads are prevented from moving circumferentially by pins between the pads, so the edge of the pad has to stay in contact with the pin, and the back of the pad has to slide against the carrier

### Pad Sliding - Pads Restrained by Centre Pins

| Pad OD | Angle | Sliding Distance |
|---|---|---|
| 89.3mm | 0.1° | 3.6μm |
| 85mm | 0.1° | 3.6μm |

For both designs, sliding distance does not vary with pad OD



## Consequences of Bearing Wear

The consequence of the wear is an increase in bearing clearance, an increased maintenance cost to replace the bearings, and a reduced level of availability if bearings have to be changed at site

The increase in bearing clearance reduces the stiffness and damping of the bearing, reducing its tolerance to unbalance and other exciting forces

In some applications, eg. Typhoon inlet bearing, this increase in bearing clearance results in an increased level of measured shaft vibration

## Degree of Bearing Wear

The maximum wear depth measured is 60μm on a pad. The wear on the carrier is generally equal to the amount on the pad. The amount of wear is variable, and depends on many factors, including:

- Number of hours and starts run
- Bearing supplier
- Which engine type and which bearing
- Position of bearing pad within the module

Other factors that may be relevant are;

- Hardness of bearing pads and carrier
- Clearance of bearing
- Surface finish of bearing and carrier
- Unbalance and vibration level

A number of examples have been examined.
The cause is considered to be fretting.

Electrical discharge pitting has been eliminated as a cause of wear
Fretting occurs when sliding surfaces wear away the oxide surface layer and expose bare metal surfaces. The sliding of these surfaces produces small particles either by repeated microwelding and breaking apart of the material surface, shearing of interlocking surface asperities or fatigue of asperities. These particles form a hard oxide ($Fe_2O_3$) and accelerate the rate of material removal
Each pad rocks with each revolution of an unbalanced shaft. The pad slides against the carrier because the whirling oil forces the pad circumferentially against the location pin as it rocks

There are a number of changes that have been made to the engines and bearings that may contribute to the faster fretting of the bearings:
- The reduction of the bearing clearance
- Use of bearings in applications where there is a higher level of synchronous and non-synchronous excitation eg. From Typhoon gearbox or Cyclone auxiliary gearbox shaft

Factors in Fretting

The single most influential factor in fretting is believed to be the amplitude of surface sliding
Secondary factors are
- material hardness
- surface finish
- contact pressure
- lubrication.

Possible causes of increased bearing fretting related to the change of supplier:

- material change. The composition of the material may be different.
- geometry change. On at least two bearing types, the <u>outer radius</u> of the pad is reduced. This causes higher contact pressure. Also the <u>bearing preload</u> and <u>circumferential length</u> may have changed. If so, these could also affect the reaction loads. The <u>method of locating</u> the pads has changed from end pins to a central pin. This may affect the amplitude of sliding.
- surface finish

## Pad Sliding

Fretting is only possible with a sliding contact between metal surfaces. It is not possible if the motion of the pads in the carrier is purely rocking.

Sliding can occur in a circumferential direction with both Glacier and Sartorius bearings

If a tilt pad was able to rock freely, the edges of the pad would move radially and circumferentially as the pad rocks. Glacier bearing pads are prevented from moving circumferentially by pins between the pads, so the edge of the pad has to stay in contact with the pin, and the back of the pad has to slide against the carrier

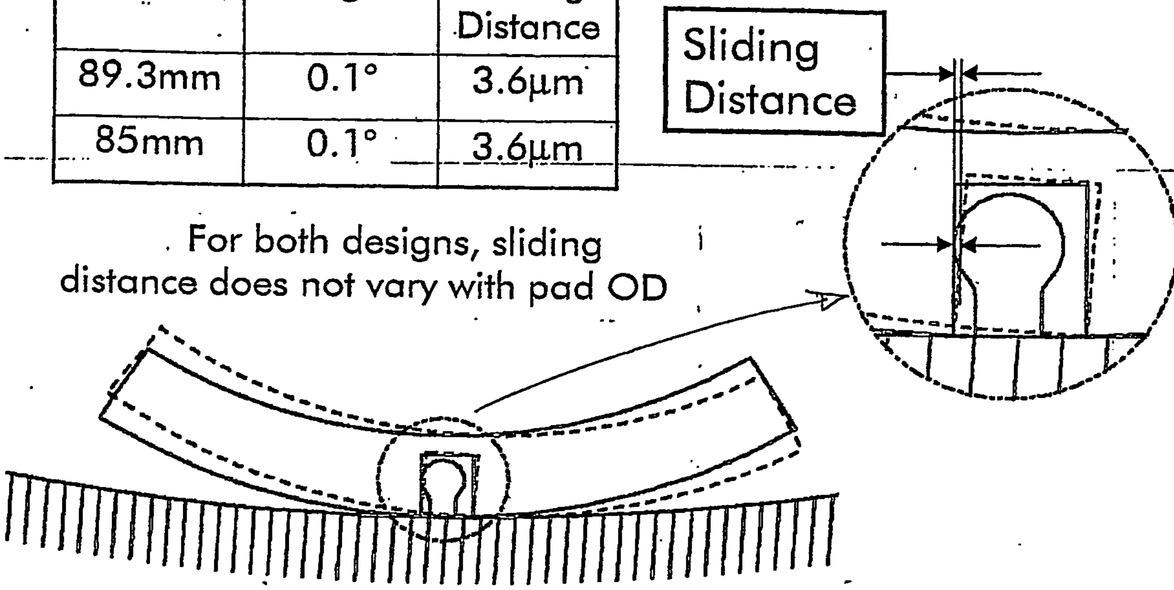
There are 2 possible reasons why a reduction in bearing clearance may cause increased fretting:

- increase in contact load. The reduced clearance bearing operates with a thinner oil film, and a higher oil film pressure. This pressure is reacted with a higher force at the back of the pad. The forced sliding of the pad will wear faster with an increased reaction load
- increase in sliding amplitude and velocity. Initial calculations indicate that the angular rocking of the pads with an unbalanced shaft is higher with a reduced clearance bearing. The sliding [illegible] will also be increased

# Pad Sliding - Pads Restrained by Centre Pins

| Pad OD | Angle | Sliding Distance |
|---|---|---|
| 89.3mm | 0.1° | 3.6μm |
| 85mm | 0.1° | 3.6μm |

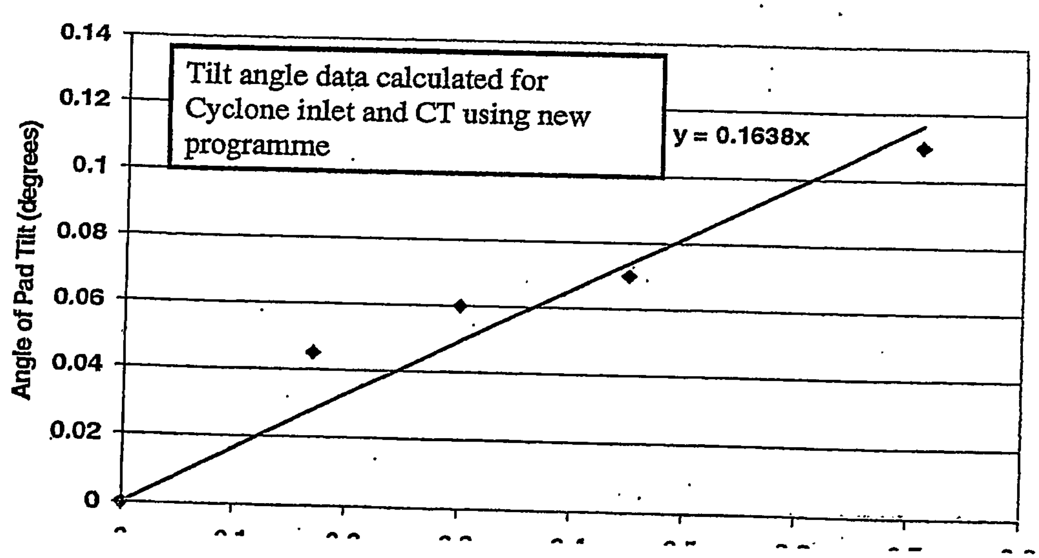
For both designs, sliding distance does not vary with pad OD




## Bearing Pad Angles

| Bearing | Unbalance | Vibration Amplitude | Bearing Clearance | Amplitude/ Clearance | Tilt Angle |
|---|---|---|---|---|---|
| Cyclone inlet | 10 kN at CT2 | 70 μm | 155 μm | 0.45 | 0.07° |
| Cyclone inlet | 20 kN at CT2 | 110 μm | 155 μm | 0.71 | 0.11° |
| Cyclone CT | 10 kN at CT2 | 53 μm | 321 μm | 0.17 | 0.045° |
| Cyclone CT | 20 kN at CT2 | 95 μm | 321 μm | 0.30 | 0.06° |

**Pad Tilt Angle v Vibration Amplitude/Clearance**



## Wear Patterns

On almost every bearing, the wear on the bottom 2 pads is significantly higher than on the other pads. The top pad almost always shows the least wear

Bearing reaction load is highest on the bottom 2 pads and lowest on the top pad

**EITHER**

**Contact load is the key parameter for bearing wear OR**

**Some other key parameter is worst on the bottom 2 pads**

NB. Hardness, surface finish cannot be worst at the bottom. Tilt angle and sliding distance may be slightly higher on the bottom pads

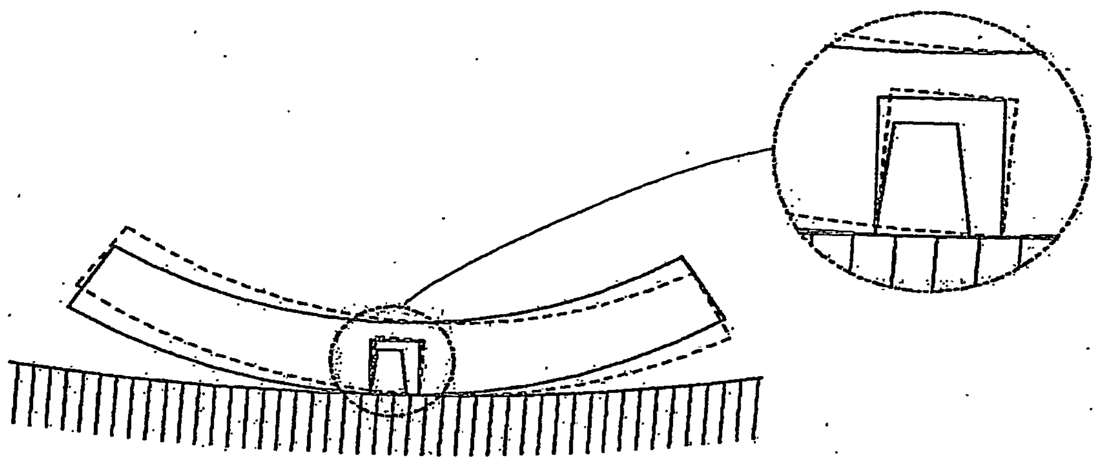
**Flutter is not the cause of bearing wear**

## Overall Conclusions So Far

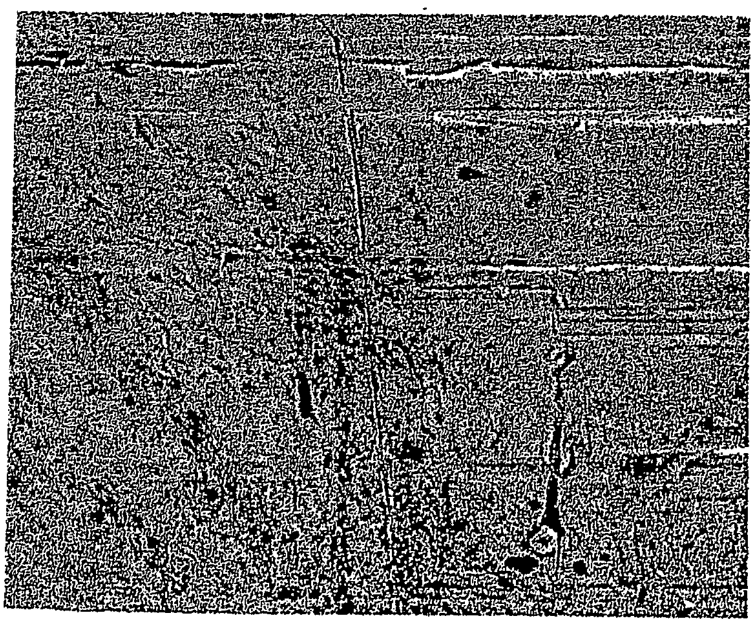
- The highest wear is seen on the most highly loaded pads
- The contact stress on the bottom pads has increased due to both a decrease in the pad radius and a reduction in clearance. This is likely to have increased the rate of wear
- The hardness of the pads has decreased
- The mounting of the pads on centre pins is likely to cause less fretting than mounting with end pins
- The fretting of centre-pin mounted pads could be reduced with conical pins

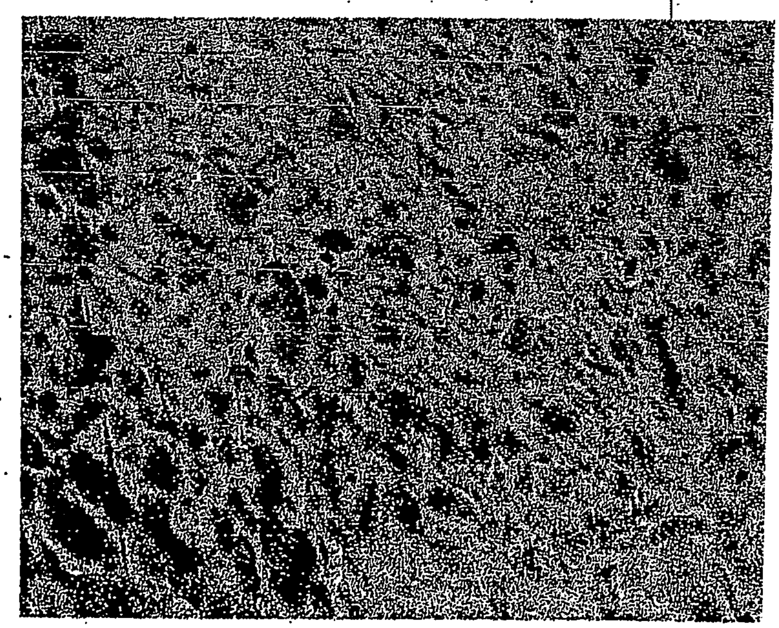
## Pad Sliding - New Centre Pin Design

Sliding Distance reduced to near zero.
Potential for fretting removed.



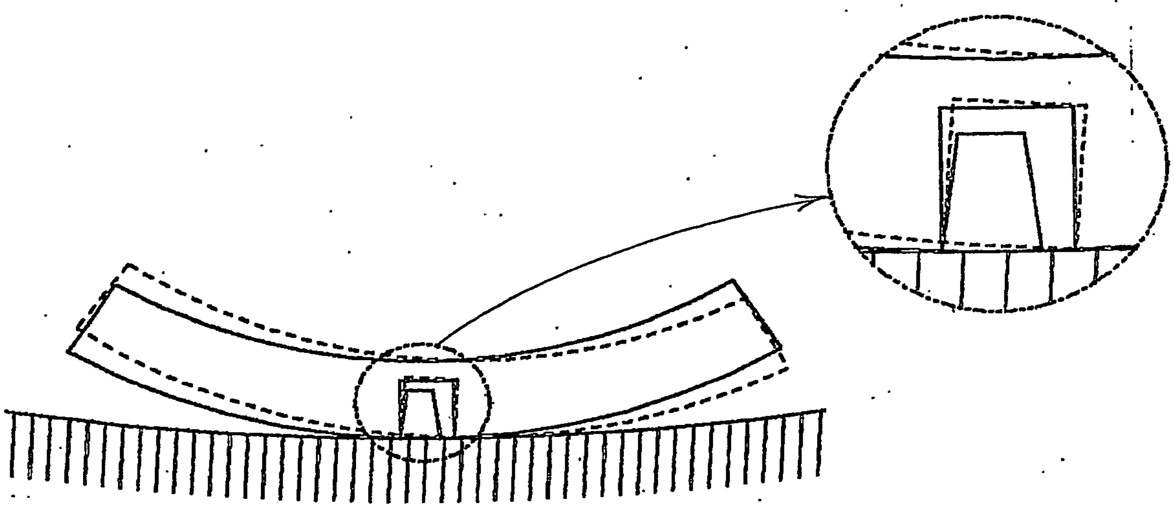
| orn -----------> Machined | Fretted Surface |
|---|---|





Pad Sliding - New Centre Pin

Sliding Distance reduced to near zero.
Potential for fretting removed.



Bearing Pad Tilt Angle and Sliding Distance

For each bearing, 60mm vibration assumed. Tilt angle calculated from slope of plot.

| Bearing | Vibration Amplitude / Clearance | Tilt Angle (Calculated) | Sliding Distance |
|---|---|---|---|
| RW Inlet (RW12031B) | 0.32 | 0.056 | |
| Cyclone/RW Inlet (MW12013) | 0.39 | 0.067 | |
| Cyclone CT/ RW CT | 0.19 | 0.032 | |
| RM Inlet (12031C) | 0.35 | 0.061 | |
| RM Inlet (12031D) | 0.44 | 0.075 | |
| RM CT | 0.26 | 0.045 | |
| RT CT (32022A) | 0.29 | 0.050 | |
| RT CT (32022B) | 0.33 | 0.057 | |

Bearing Pad Contact Stress

RM Inlet Bearing, RM12013, 12031-12031D

Contact stress between pad backing and carrier for a perfectly balanced rotor (static loading)

| | Glacier (013) | | Sartorius (31C) | | Sartorius (31D) | | |
|---|---|---|---|---|---|---|---|
| | Original Clearance | | Original Clearance | | Reduced Clearance | | |
| | Top pad | Bottom | Top pad | Bottom | Top pad | Bottom | |
| Pad reaction load = | 123 | 1126 | 123 | 1126 | 397 | 1422 | N |
| Pad outer diameter $D_2$ = | 89.3 | 89.3 | 85.0 | 85.0 | 85.0 | 85.0 | mm |
| Contact stress = | [redacted] | [redacted] | [redacted] | [redacted] | [redacted] | [redacted] | MPa |

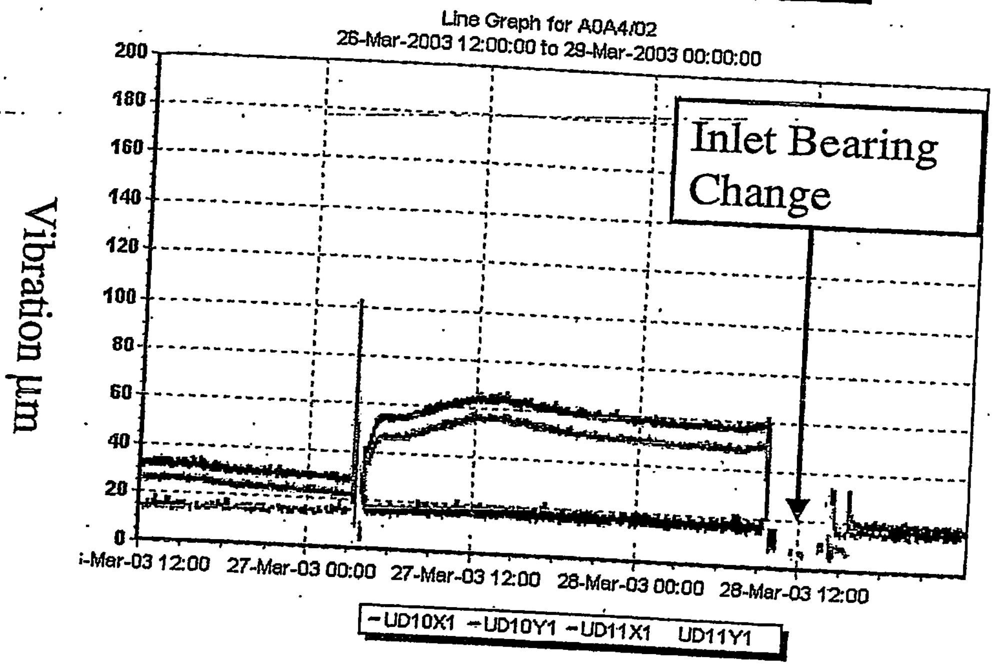


RM Inlet Bearing, RM12013, 12031-12031D

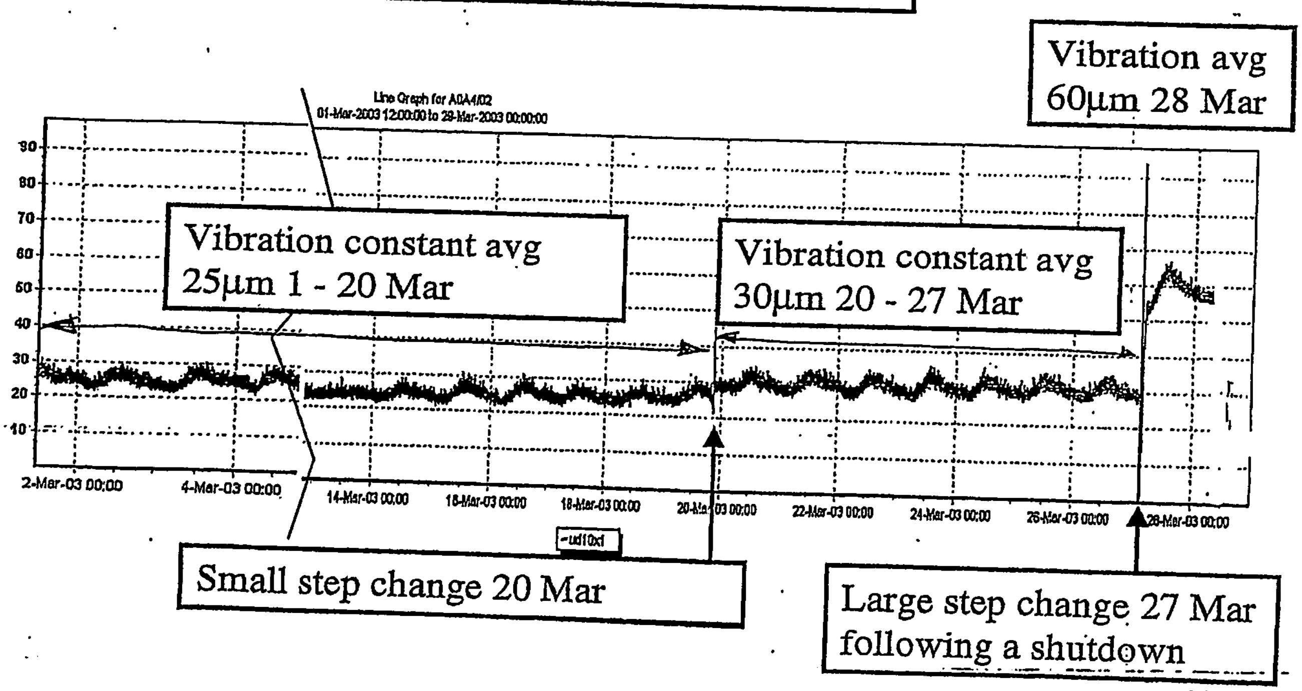
Additional dynamic contact stress from 1000gmm unbalance on the inlet bearing

| | Glacier (013) | | Sartorius (31C) | | Sartorius (31D) | | |
|---|---|---|---|---|---|---|---|
| | Original Clearance | | Original Clearance | | Reduced Clearance | | |
| | Top pad | Bottom | Top pad | Bottom | Top pad | Bottom | |
| Pad reaction load = | 1525 | 1525 | 1525 | 1525 | 2146 | 2146 | N |
| Pad outer diameter $D_2$ = | 89.3 | 89.3 | 85.0 | 85.0 | 85.0 | 85.0 | mm |
| Contact stress = | 33 | 33 | 44 | 44 | 51 | 51 | MPa |
| Total contact stress = (static + dynamic) | [redacted] | [redacted] | [redacted] | [redacted] | [redacted] | [redacted] | MPa |

# Vibration on Bulwer Cyclone



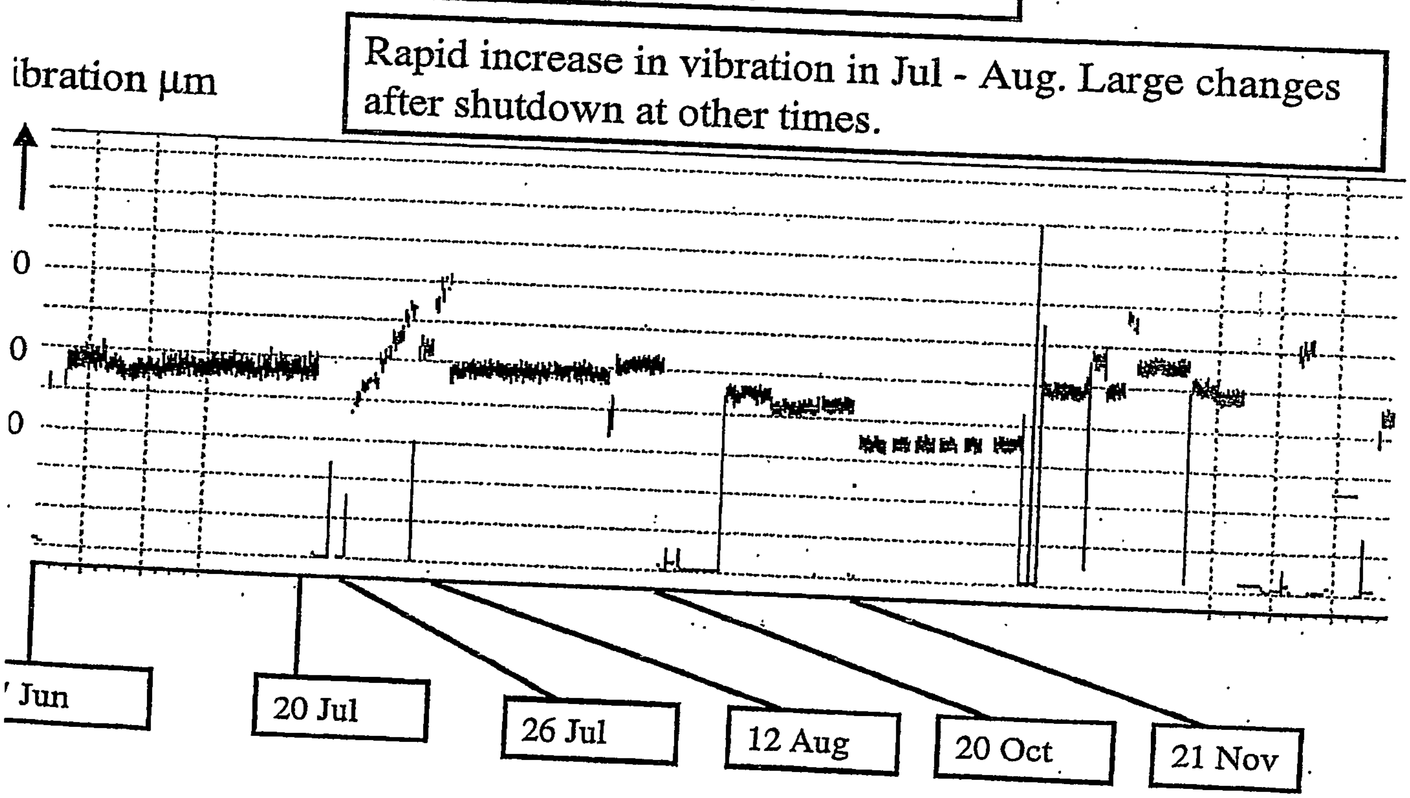

# Vibration on Bulwer Cyclone



# Vibration on Bulwer Cyclone - Conclusions

- ...n does not increase steadily with time, but in 2 ...tep changes within March 2003
- ...earing wear cannot occur in steps
- ...earing clearance must have increased steadily
- **...earing wear is not the cause of the vibration increase**
- **...ibration level on the Cyclone inlet bearing is not ...nsitive to bearing clearance**
- ...argest increase in vibration occurs after a shutdown
- **...hanges during a shutdown could be a cause of ...bration**

## Vibration Carberry Milk Typhoon




A change in July caused the vibration to start increasing. This could be due to bearing pad wear

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